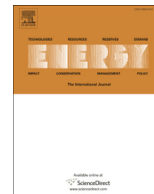




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Travelling-wave thermoacoustic high-temperature heat pump for industrial waste heat recovery

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ABSTRACT

Many industrial processes need steam at temperatures from 100 to 200 °C, normally produced by directly heating water via coal, natural gas or oil combustion. Nevertheless, large amounts of unused heat below 100 °C are wasted in other industrial processes. In principle, a high-temperature heat pump capable of using the industrial waste heat can provide steam above 100 °C. However, until now, efficient and reliable heat pump technology for the application is not available. In this paper, a novel TWTAHP (travelling-wave thermoacoustic heat pump) is presented to meet this requirement, which can potentially solve the problems occurring in conventional vapour-compression heat pump such as high discharge temperatures, high pressure ratio, and low efficiency. This system comprises three linear pressure wave generators which are coupled with three heat pumps into one single closed loop. Theoretically, this system is able to complete the thermoacoustic conversion with a much higher efficiency. The theoretical simulations were performed at varied waste-heat temperatures (40–70 °C) and different hot-end temperatures (120–150 °C). The computing results show that this new heat pump system has a high relative Carnot efficiency of about 50%–60%. In using a reliable linear compressor and a thermoacoustic heat pump with no-moving parts, this technology has an inherent potential for high reliability. Therefore, it is believed that the travelling-wave thermoacoustic heat pump is an enabling technology with good prospects in efficiently harvesting industrial waste heat.

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1. Introduction

As the depletion of Earth's energy resources continues to be a serious problem, saving energy and improving energy using efficiency become imperative. Heat pump systems offer economical alternatives in recovering waste heat from different sources for using in various industrial, commercial and residential applications. With the rising of fuel costs and global warming at the forefront of the world's attention, the interest in heat pumps as a means to recover energy has grown. However, such systems have not been applied as widely as they should or could be. In traditional heat pump systems on the basis of the vapour compression cycle, system design and optimisation remain challenging problems as rising the delivering temperature can result in higher discharge temperature,

higher pressure ratio, and lower performance efficiency. On the one hand, there are huge amounts of waste heat below 100 °C from industrial processes that have not to be used yet. On the other hand, many processes need steam at the temperatures higher than 100 °C, which is currently produced by directly heating water with the combustion of coal, natural gas, or oil. It is urgent that an efficacious method should be found to transform the low-grade heat sources into the high-grade heat sources. Wang [1] carried out theoretical and experimental investigations on a moderately high-temperature heat pump using environmentally-friendly refrigerants. The result indicated that on the evaporating and condensing temperatures of 44 °C and 90 °C, this heat pump could achieve an associated COP_h (coefficient of performance with heat pump) for HC600 and HC600a of 3.84 and 3.33 respectively. Using a high-temperature heat pump with the zeotropic mixtures as refrigerants, Zhao [2] studied the nonlinear relationship of various system parameters. The theoretical results showed that the zeotropic mixtures, which have a smaller maximum temperature difference in the condenser and a smaller minimum temperature difference in the evaporator could attain the highest COP.

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Antonić [3] studied six different heating systems on the basis of the vapour compression cycle with either two-stage or one-stage compression. The two-stage compression had the best seasonal COP as well as the system exergy parameters, which can attain a COP of nearly 4.0 with the evaporating and condensing temperatures of nearly -10°C and 65°C respectively. In the recent research, the traditional vapour compression system has been advanced in many different areas. However, despite some valuable improvements, there would be an increased cost associated with using more complicated and expensive hermetic reciprocating compressors. Dong Ho Kim et al. [4] set up a two-stage air–water source heat pump, using R134a and R410A as refrigerants based on prophase simulation and optimization calculation. The experimental results showed that at the operating condition of the ambient temperature at -7°C and the heating capacity with 15 kW, a downward trend in COP could be obtained with the increased demand for hot water temperature. When the hot water temperature increased to 55°C , the COP dropped to 2.0 or less, while reducing the ambient temperature also caused the same problem. Wei Yang et al. [5] designed a direct-expansion ground source heat pump in Xiangtan, China for comparing with the traditional ground source heat pump. With the operating condition of the ambient temperature at 4.8°C , 13.5°C and the heating temperature at 50°C , the new system could obtain the heating COP of 4.73 in average. M. Noro et al. [6] made a research of a multisource heat pump system, which included evaluation and analysis of the data obtained through real time monitoring of the working system in operation, for a period of approximately two heating seasons. During this time, the behaviour of the system was assessed and the incorrect settings of the plant were identified and subsequently adjusted as required. The energy balance indicated that the integration of different sources not only increased the thermal performance of the system as a whole, but also optimized the usage of each source.

In this paper, a novel TWT AHP (travelling-wave thermoacoustic heat pump) is presented to meet the requirement, which could possibly solve the problems occurring in conventional vapour compression heat pumps. Thermoacoustically driven systems for heating or cooling have received much attention in recent years for

their heat-driven mechanism, without moving parts, and structural simplicity. Thermoacoustic machines, which mainly include thermoacoustic heat engines and refrigerators, make use of thermoacoustic effect to realize the conversion between heat and sound energy. Until 1999, Swift et al. [7] studied a pulse tube refrigerator with acoustic power recovery of lost power in the orifice-type pulse tube cryocoolers, which is now called thermoacoustic-Stirling refrigerator or TWTAR (travelling wave thermoacoustic refrigerator). Recently, much effort has been done in developing TWTARs for higher cooling temperature range [8,9], in which the linear compressors are more widely used as the drivers. Taking advantage of thermoacoustic theory, we have designed a TWT AHP system to test its effectiveness. The theoretical model, the optimisation procedure and the results of this new system will be introduced.

2. Theoretical model

The TWT AHP (see schematic in Fig. 1) composes three linear pressure wave generators coupled with three heat pump sections in one closed loop. Each heat pump section (see Fig. 2) includes an ambient heat exchanger, thermal buffer tube, hot-end heat exchanger, regenerator, cold-end heat exchanger, and connecting tubes. All the heat exchangers are made of copper with the rest of the elements are made of stainless steel. The linear pressure wave generator is based on a dual-opposed piston design that minimises vibrations. By adding a little amount of the electrical power into the generator, the acoustic power will be produced by the generator and transfer from left to right through the helium as the medium in every heat pump section, as depicted in Fig. 2. The oscillation of sound in the gas medium can make the change of temperature inside the molecules of gases, which mainly happens in the regenerator. The regenerator is the core of this system, which is filled with pieces of the stainless steel wire mesh. The gas in this porous medium has lower fluidity and higher heat conduction. As a result, the two ends of the regenerator will have the highest temperature and the lowest temperature respectively. Then, with the thermoacoustic conversion in regenerator, the cold-end heat exchangers begins to absorb heat from the waste heat source and the hot-end heat exchangers started to deliver heat to the user area. The thermal buffer tube is used to decrease and control the temperature in ambient heat exchanger. Then the ambient heat exchangers could protect the linear pressure wave generators from over heat. With this structural arrangement, when the acoustic power is transferred to the next heat pump section, the phase angle for the volume flow rate decreases 120° , which is one of the best phase in travelling wave system. At last, the transmission of sound wave will come back to the beginning. In this way, the system is able to use low grade heat sources to obtain heat energy at much higher temperatures.

Thermoacoustic theory is a means to understand the working mechanism of a TWT AHP. The model that we employed in designing our heat pump is based on this theory which was

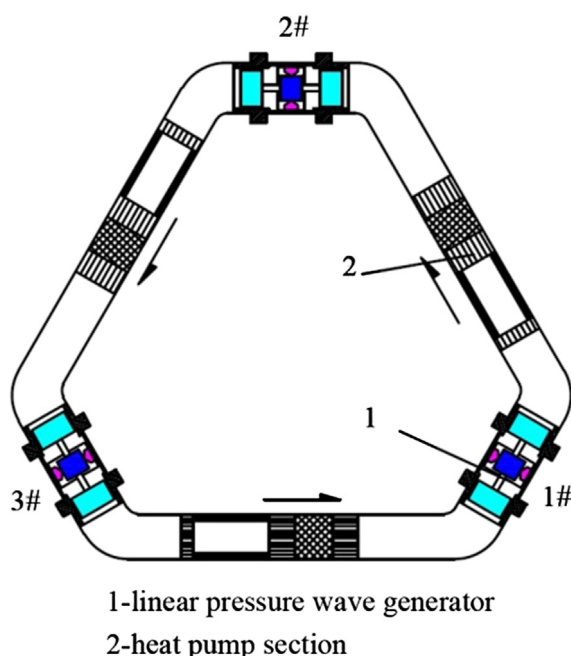


Fig. 1. Schematic of the travelling-wave thermoacoustic heat pump.

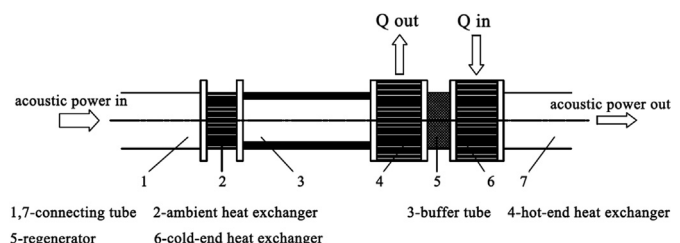


Fig. 2. Detail of one heat pump section.

developed by Rott [10]. All the calculations are based on the laminar flow hypothesis. Accordingly [10,11], the thermoacoustic versions of the momentum, continuity, and energy calculation equations are as follows:

$$\frac{dp_1}{dx} = -\frac{i\omega\rho_m}{A}U_1, \quad (1)$$

$$\frac{dU_1}{dx} = -\frac{i\omega A}{\rho_m a^2}p_1, \quad (2)$$

$$\frac{dH}{dx} = -Q, \quad (3)$$

$$\dot{E} = \frac{1}{2}\text{Re}[p_1\tilde{U}_1]. \quad (4)$$

The Carnot efficiency, the COP_h (which is identified as the proportion of heating capacity and consuming acoustic power in the system), and the relative Carnot efficiency (which is identified as the proportion of COP_h and Carnot efficiency to see the thermodynamic perfect degree) in thermoacoustic system are defined by:

$$\eta_c = \frac{T_h}{T_h - T_c}, \quad (5)$$

$$\text{COP}_h = \frac{Q_h}{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}, \quad (6)$$

$$\eta_{\text{RC}} = \frac{\text{COP}_h}{\eta_c}. \quad (7)$$

Combining this form of the equations of all the computational cells leads to a sparse matrix which can be easily solved using linear algebra algorithms. For this paper, the numerical model was implemented using DeltaEC software developed by Los Alamos National Laboratory [12].

For the numerical simulations, the mean pressure is set at 5 MPa, the frequency at 80 Hz, the cold-end temperatures are between 40 and 70 °C which come from the waste-heat in industry progress, and the hot-end temperatures are between 120 and 150 °C. The parameters of the linear pressure wave generator are listed in Table 1.

Given the specifications of the linear pressure wave generator, the physical dimensions of the heat pump section were determined to achieve optimal operational efficiency. The influence of the core elements, include heat exchangers and regenerator, was considered as well. Finally, system performances at different cold-end and/or hot-end temperatures were also investigated.

3. Numerical results

3.1. Characteristics and structure optimisation of TWAHP system

We analysed the distribution of the acoustic power and energy flow in a completed TWAHP system working at 70 °C to 150 °C with the existing structural components (see Fig. 3). The linear

Table 1
Linear pressure wave generator parameters.

Nominal voltage V	Piston diameter mm	Amplitude mm	Moving mass kg	Stiffness kN/m
416	75	6.5	2.3–3.5	130

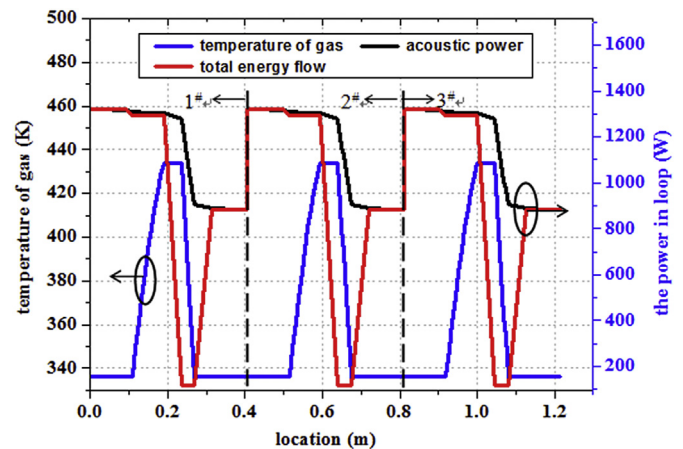


Fig. 3. Distribution of energy flow in a TWAHP system working at 70 to 150 °C.

pressure wave generator works as a compressor at the start of each heat pump section and as an expander at the outlet of each preceding heat pump section. The acoustic power is produced in the compression chamber and assimilated by the heat pump section; the rest of the acoustic power is recovered by the expansion chamber in the next generator. The results show that nearly 30% of the consumed acoustic power is taken up by the regenerator in the heat pump section; the rest can be used to keep the cycle working continuously and stably.

According to this original model, we keep optimising the TWAHP structure, especially in the heat pump section. Our main ultimate objective is to maximise COP_h ; Table 2 has listed the results for structure optimisation within the heat pump section.

Based on the above structure, we can obtain a maximal COP_h of 2.99 with 1038.7 W heating capacity and 713.44 W heat absorbing capacity in each heat pump section, consuming only 347.31 W acoustic power at the working condition mentioned in Fig. 3.

3.2. The influence of regenerator and heat exchanger

To provide steady and efficient operating conditions, the regenerator and the two main heat exchangers (hot-end and cold-end heat exchangers) are the more crucial components in the whole system. Firstly, we have studied the regenerator, which is a stainless-steel cylinder filled with stainless-steel wire mesh with mesh number of 150. The transformation between acoustic power and heat is produced inside the regenerator which can affect the performance of the whole system substantially. Fig. 4 shows the results of calculations, for which we have changed only the length of the regenerator.

Table 2
Optimisation structure of the heat pump section.

Components	D mm	L mm	Parameter
Connecting tube 1 [#]	50	104	
Ambient heat exchanger	50	20	Copper shell-tube exchanger, $d = 1$ mm, $\phi = 0.25$
Buffer tube	50	80	Wall thickness is 2.5 mm
Hot-end heat exchanger	60	34	Copper shell-tube exchanger, $d = 1$ mm, $\phi = 0.25$
Regenerator	60	25	Wire mesh fills, $D_{\text{wire}} = 0.066$ mm, mesh number is 150
Cold-end heat exchanger	60	28	Copper shell-tube exchanger, $d = 1$ mm, $\phi = 0.25$
Connecting tube 2 [#]	50	140	

The superscript “#” has the same meaning with “number”.

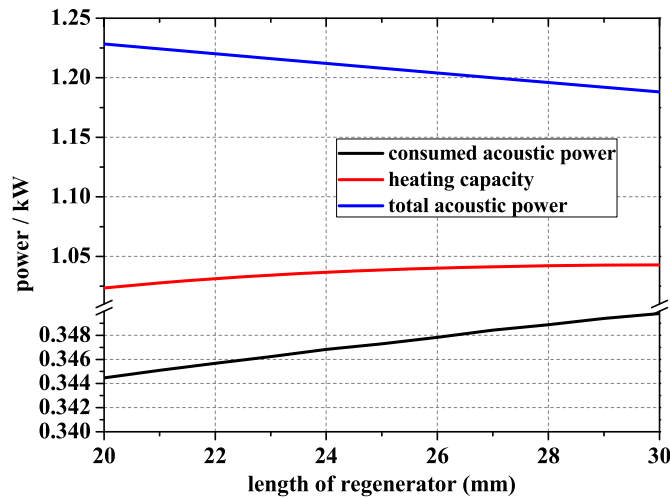


Fig. 4. Variation in performance with different length regenerators.

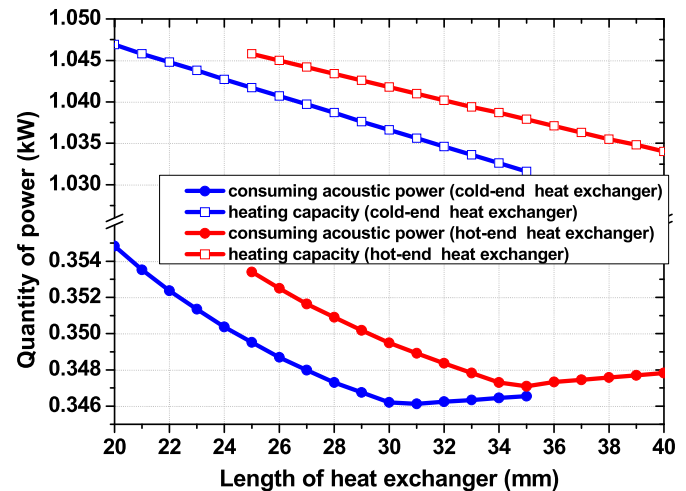


Fig. 6. Variation in energy conversion with heat exchanger length.

There is a maximal point for which a COP_h of 2.99 with 56.55% relative Carnot efficiency has been obtained. The change in the total acoustic power is more obvious than for the other parameters, which clearly decreases with the increasing regenerator length. It is accepted that the heating capacity increases slowly with the rising length because more acoustic power is being consumed.

Connecting to the two ends of the regenerator are two important heat exchangers, a hot-end heat exchanger and a cold-end heat exchanger following the direction of input acoustic power. The geometric structure, heat-transfer type and performance of these heat exchangers should better match the designed regenerator and the linear pressure wave generator in order to make the whole system obtain a reasonable result. Fig. 5 shows the COP_h variation with the length of heat exchangers.

To maintain a much higher operating performance and heat transfer efficiency, we chose shell-tube copper exchangers combined with the designed regenerator. To achieve a maximal COP_h , we have estimated the optimal length for the hot-end heat exchanger and the cold-end heat exchanger from Fig. 5. Fig. 6 shows the energy conversion variation with increasing the heat-exchanger length. Heating capacity and consuming acoustic power are the most important variables that we concern about. The

numerical results show that there is an absolute minimum in the consumed acoustic power in both heat exchangers corresponding to a maximum COP_h . In contrast, the heating capacity in both heat exchangers declines with increasing the heat-exchanger length.

3.3. The influence of operating temperature

Using the optimised components, the heat pump sections in the numerical simulations are able to be connected to the linear pressure wave generators. By increasing the cold-end temperature from 40 °C to 70 °C (keeping hot-end temperature at 150 °C), the COP_h and relative Carnot efficiency curves are plotted (Fig. 7). As cold-end temperature increases, the COP_h increases as well, but the relative Carnot efficiency declines. The possible reason could be a quicker increase in the Carnot efficiency as the temperature difference is shrunk.

Fig. 8 shows the variation of performance parameters with increasing cold-end temperature. Both heating capacity and heat absorbing capacity (the heating absorbed from the waste heat in cold-end heat exchanger) increase slowly, whereas the consumed acoustic power declines, which could explain the reason for a higher efficiency at a much higher cold-end temperature.

Fig. 9 shows the dependence of COP_h and relative Carnot efficiency on the hot-end temperature in the range between 120 °C

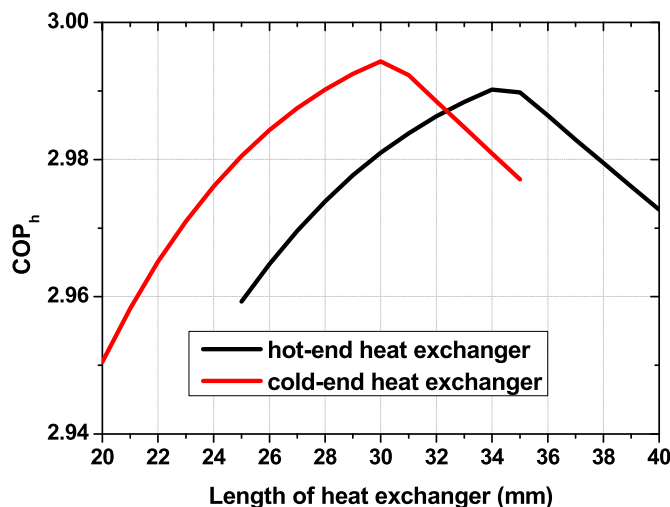


Fig. 5. Variation of COP with heat exchanger length.

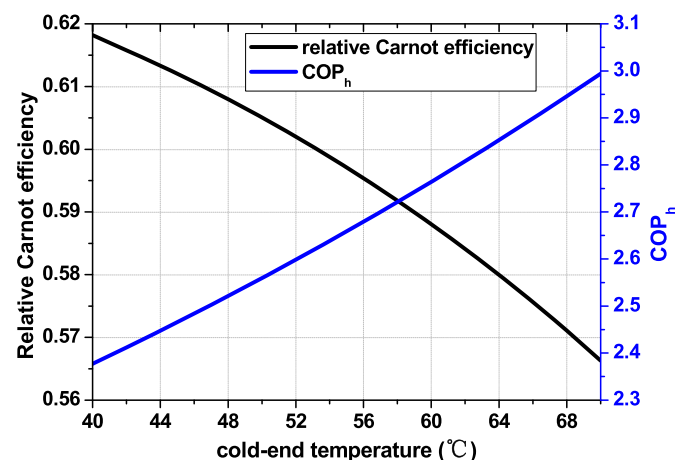


Fig. 7. Efficiency curve with changing cold-end temperature.

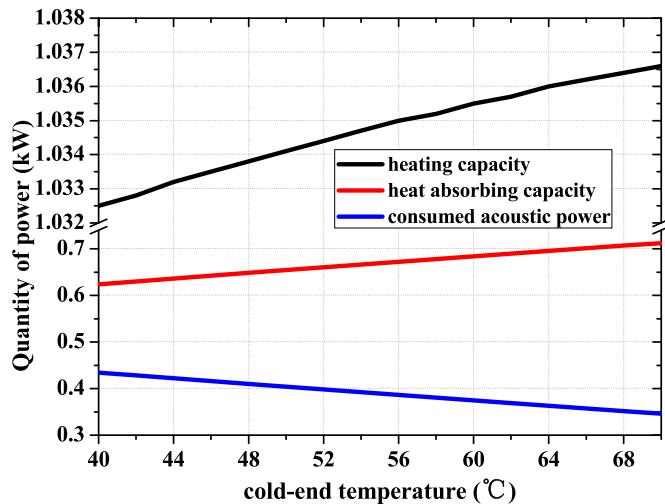


Fig. 8. Dependence of performance curves with changing cold-end temperature.

and 150 °C while maintaining the cold-end temperature at 70 °C. As the hot-end temperature rises, the curves show contrary trends compared with those in Fig. 7; the best operating condition could be achieved the lowest hot-end temperature in this paper.

Accordingly, Fig. 10 shows the variation of performance parameters with increasing the hot-end temperature. As both the hot-end temperature and the temperature difference increase, operating performances become worse. There is a perceptible reducing in heat absorbing capacity while more acoustic power is demanded by the operating. During the change of temperature, heating capacity has not been significantly affected.

To make a summary, by using an optimal structure, the bigger temperature difference, the lower system performance. It seems that the law is much similar as in traditional vapour compression system. However, the advantage with this new system is that the COP_h can get to 3.0 even at 150 °C in the hot-end heat exchanger with the temperature difference increases to 80 °C. By the constraint of compressor and refrigerant, when the hot-end temperature rises, the COP_h will drop rapidly below 3.0 in traditional system without improvement. At the opening, some references [1,3,6] were given to review the improved research. As we can see, through two-stage system and new type refrigerant are used in traditional heat pump for upgrade the efficiency, these efforts have had little success. In the other hand, this new system is totally

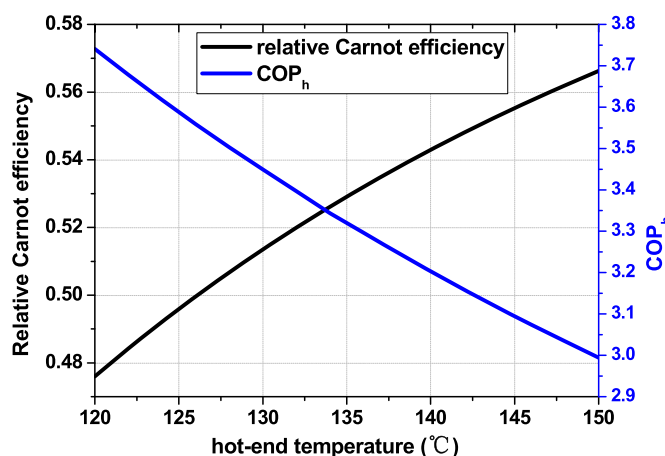


Fig. 9. Efficiency curve with changing hot-end temperature.

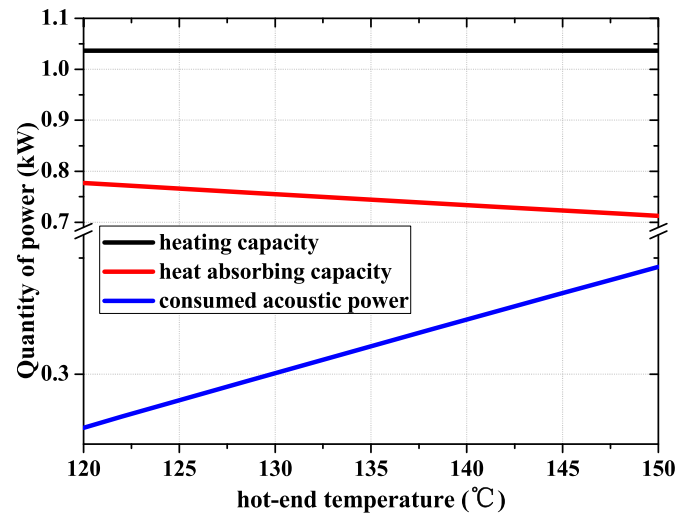


Fig. 10. Dependence of performance curves with changing hot-end temperature.

different from the traditional heat pump, which will cause the economic cost in producing and installation. At the very beginning, the new one will be more expensive than using the old one. However, the potential in the efficient harvesting of industrial waste heat could counterbalance the price in the nearly future.

4. Conclusions

In this paper, a novel TWAHP system was designed, simulated and optimised based on thermoacoustic theory. We believe that the TWAHP is an enabling technology and has good potential in the efficient harvesting of industrial waste heat.

- (1) The optimal structure of the heat pump section has been obtained by simulations using the DeltaEC program. With this optimal structure, one heat pump section can attain a COP_h of 2.99 with 1038.7 W heating capacity and 713.44 W heat absorbing capacity consuming only 347.31 W of acoustic power at 70 to 150 °C.
- (2) In all conditions, we calculated that each heat pump section can offer more than 1000 W heating capacity. Were an operationally compatible linear pressure wave generator to be used, the COP_h and the heating capacity would certainly increase.
- (3) An increase in the temperature difference could have a negative influence on system performance, which is of similar order as encountered in traditional vapour compression systems. Note though the effect is much slighter than for the traditional systems.

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Nomenclature

A	cross-sectional area, m^2
a	sound speed, m/s
COP	coefficient of performance

D, d	diameter, m
$\dot{E}$	acoustic power, W
f	frequency, Hz
H	total energy flow, W
I	electric current, amp
i	$\sqrt{-1}$
L	electric inductance, Henry
L, l	length, m
M	mass, kg
m	screen mesh number, wires/m
p	pressure, MPa
Q	quantity of heat, W
R	electric resistance, kg/s
$\text{Re}[]$	real part of
T	temperature, K
U	volume flow rate, m ³ /s
x	displacement or location, m
Z	electric impedance, ohm

Greek symbols

ω	radian frequency ($=2\pi f$), s ⁻¹
η	efficiency
ϕ	volumetric porosity
ρ	density, kg/m ³

Subscripts

1	first order, usually a complex amplitude
C	Carnot
c	cold end
h	hot end or heat pump

m	mean
min	minimum
max	maximum
RC	relative of Carnot
tot	total
κ	thermal
ν	viscous

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